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## Low Temperature Expansion Turbines

### Turbines à expansion à basses températures

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**SOMMAIRE.** Le développement des grandes installations pour la production d'oxygène gazeux est inséparablement lié à la construction d'une turbine à expansion sûre et simple travaillant à basses températures. Une turbine à expansion réactive du type centripète dont le rendement adiabatique était de 80% fut développée en 1939 sous la direction de l'académicien P. L. Kapitza. La différence principale entre ce nouveau type de turbine à expansion et la turbine à expansion de Linde du type actif est quoique la grille de l'"impeller" du premier est du type "confusor", que l'expansion du gaz a lieu dans l'"impeller" grâce au gradient de pression considérable créé par l'action des forces centrifuges dans le "long blade impeller". Des expériences ont montré que les exigences principales pour atteindre un haut coefficient de performance consistent en la "confusional mode" du courant de gaz dans l'"impeller", les vitesses basses absolues du gaz quittant l'"impeller", l'usage correcte du gradient de pression formé par l'action des forces centrifuges et la création de conditions pour lesquelles la vitesse du courant du gaz venant des "guide vanes" ne dépasse pas la valeur critique d'une façon importante. Des projets pour des turbines à expansion réactives à basses températures ont été faites pour des capacités de 500 à 20.000 Nm<sup>3</sup>/hr d'air. Le rendement adiabatique de grandes machines peut aller jusqu'à 82-83%. Les résultats des recherches démontrent la possibilité d'augmenter encore le rendement. Les turbines sont à un étage, avec soit un "impeller" bilatéral monté sur un axe flexible amorti, soit un "impeller" unilatéral cantilever monté sur un axe rigide. Plusieurs projets de turbines à expansion sont discutés dans cette communication et des recommandations sont données en ce qui concerne les relations et paramètres optimaux.

The last decade has been characterized by the extensive application of gaseous oxygen and other products of air separation in different fields of industry in many countries. This trend is accompanied, of course, by the development of new equipment for large-scale low temperature air-separation plants.

Big modern air-separation plants for obtaining gaseous products operate on low-pressure air refrigeration cycles using compressors and expansion engines of the rotary type. Work on low-pressure air-separation installations was started in the U.S.S.R. under P. L. Kapitza on the basis of his new expansion turbine. The high isentropic efficiency peculiar to this new expansion engine which, in the low-pressure air-separation installation, is the only source for refrigeration, ensured the development of large-scale oxygen production in big and economical air-separation plants. Installations producing up to 15,000 nm<sup>3</sup> oxygen per hour per unit, working continuously for two years, are being successfully run in the U.S.S.R. They have low temperature inward radial flow reaction turbines developed on the basis of that proposed by P. L. Kapitza. At present expansion turbines of this type have been developed in various countries. J. Wucherer pointed out in a communication at the Congress that such expansion turbines are reliable in operation and notable for their high efficiency.

The experience acquired in designing, investigating and operating expansion turbines at low temperatures enables us to put forward a contribution on this subject. We hope that discussion on the design and construction of expansion turbines will be of great use and favour their further development.

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Unlike the turboexpanders of the impulse type developed formerly by the Linde Company, in the inward radial flow reaction turbine the air is expanded not only in the stationary nozzles but also in the channels of the wheel at a constant or accelerated channel velocity. This is achieved by using wheels with "long blades",

i. e. with a small ratio of exit to entrance diameter of the wheel  $\frac{d_2}{d_1}$ , which creates a substantial pressure gradient due to the effect of the centrifugal forces. Under the operating conditions of the low-temperature expansion turbine designed in air-separation units for a comparatively small enthalpy drop (about 25 kcal/mole), such a construction made it possible to lower considerably the energy losses due to the flowing of the gas stream across the turbine. Thus it was possible to keep the gas discharge velocity from the stationary nozzles about the sonic and to employ simple nozzles. Accordingly the energy losses in the nozzles and in the radial clearance between the nozzle box and the wheel were reduced. Thanks to the small diameter ratio  $\frac{d_2}{d_1}$  the change in the direction of the gas stream is smooth and not very large. This, together with the small relative velocities of the gas stream at the inlet and in the course of its usually accelerated passage along the channels of the wheel, led to a reduction of losses across the wheel. The gas stream leaves the blades at a comparatively small velocity due to the small ratio of  $\frac{d_2}{d_1}$ ; this consequently cuts down the losses dependent on the exit velocity. These are the principal features which made it possible to obtain the high isentropic efficiency peculiar to the inward radial flow reaction expansion turbines. However, as compared with the impulse turbines, the losses due to the disk friction and flowing of the gas past the blades are considerably higher in reaction turbines where the pressure drop across the wheel is substantial and the gas in the space between the wheel and the casing has a higher pressure.

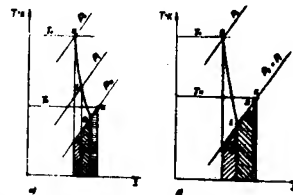


Fig. 1. T-S diagram of the gas expansion process. a) reaction expansion turbine. b) impulse turbine. Refrigerative losses: I, in the stationary nozzles, II, across the wheel and the discharge velocity, III, due to leakage through the inner labyrinth glands.

Fig. 1 shows the T-S diagram of the process for the reaction expansion turbine and impulse turbine; the main refrigerative losses are indicated respectively. Refrigerative losses are energy losses determined at the terminal expansion pressure by the product of the entropy increase and corresponding mean temperature. The different states of the gas are indicated as follows: 0 - the initial state, 1 - before the wheel, 2 - behind the wheel, K - the terminal state taking into account the changes in the state of the gas caused by contact with the part of the gas which had passed through the inner labyrinth glands.

If the design and construction ratios are selected properly the total refrigerative losses are considerably smaller in the inward radial flow reaction turbines than in impulse turbines.

Low-temperature expansion turbines are being operated in the U.S.S.R. handling from 500 to 20,000 nm<sup>3</sup> air per hour. The isentropic efficiency at liquefying temperatures is 82-83%. J. Wucherer reported even higher efficiencies. It should be mentioned, however, that in evaluating the efficiency of expansion turbines

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which operate at a discharge gas temperature close to boiling point and at small enthalpy drop, the accuracy in determining the efficiency depends on the state diagrams used and even on the method of calculating the enthalpy by means of these diagrams. The calculation of efficiency is also affected by the method of determining the experimental data, *i.e.* the pressure and, particularly, temperature of the gas which changes considerably in time due to the switching over of the regenerators.

As a result of investigations and experience in running expansion turbines the following optimal design data for low-temperature single-stage expansion turbines have been found:

$$\text{Exit to entrance diameter ratio} \quad \frac{d_2}{d_1} = 0.38 \div 0.45$$

$$\text{Dimensionless wheel inlet width} \quad \frac{b_1}{d_1} = 0.025 \div 0.03$$

$$\text{Ratio of the peripheral wheel speed to the theoretical spouting velocity which corresponds to the total isentropic enthalpy drop} \quad \frac{U_1}{C_0} = 0.62 - 0.67$$

$$\text{Reaction, i.e. the relation of the isentropic enthalpy drop across the wheel to the total enthalpy drop} \quad q = 0.35 \div 0.42$$

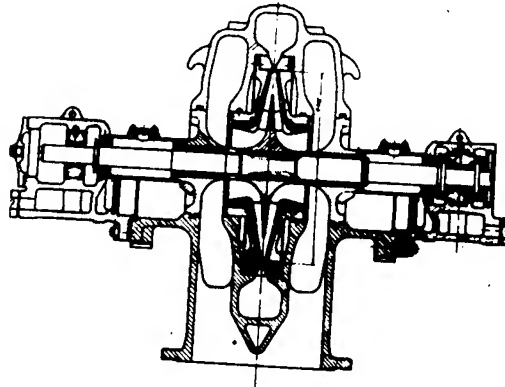


Fig. 2. Inward radial flow reaction turbine with a two-exits-wheel.

In Fig. 2 and 3 two types of reaction expansion turbines for air-separation units are shown. These turbines are single-stage, quite reliable engines, simple in design. These two types differ in the design of their wheels. Figure 2 shows an expansion turbine which has a wheel with two exits on a flexible shaft. Fig. 3 shows one possible to design wheels of smaller diameter and consequently slightly lower the losses due to disk friction and leakage of gas through the labyrinth glands, as well as to obtain a more favourable dimensionless wheel inlet width and have the rotor free from the influence of axial forces. However, in this case the employment of an elastic damping device is required. As experience in running air-separation units has shown, in expansion turbines on a flexible shaft without a special elastic bearing the dynamic steadiness of the rotor is not ensured in cases of disturbances when the liquid air might get into the turbine.

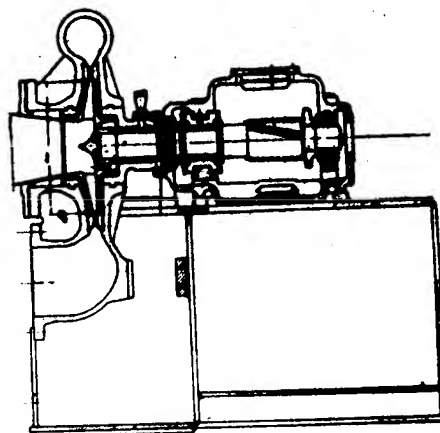


Fig. 3. Inward radial flow reaction turbine with a one-exit-wheel.

A bearing with a damping device as it is shown in Fig. 4, secures the dynamic steadiness of the turbine rotor.

The expansion turbine with a wheel which has one exit is simpler. The bearing arrangement of the high-speed shaft of the gear box gives cantilever support of the turbine wheel. To balance the axial forces, the labyrinth glands are usually located on both sides of the wheel at about the same diameter, and several drillings bored through the main wheel disk close to the hub.

Labyrinth glands with sufficiently numerous combs are employed to reduce to 3-4% the refrigerative losses due to the flowing of the gas past the blades. The leakage of cold gas through the labyrinth glands on the shaft is usually less than 1%.

Comparing these two construction types one might have expected the isentropic efficiency of the turbine with a two-exits-wheel to be higher than of that with a one-exit-wheel designed for the same conditions. As experience has shown this difference is not very essential. Thus the manufacture of expansion turbines with a one-exit-wheel is fully justified because of their notable simplicity in construction and reliability in operation.

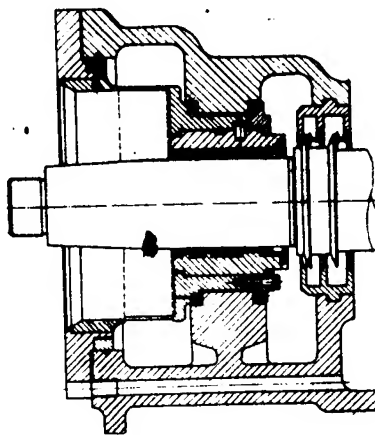


Fig. 4. Elastic damping bearing.

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The blade arrangement may be of numerous thin blades (in which case half of them are shortened and located only at the entrance) as in our first engines, or it may be of the inverse centrifugal compressor type, and then the blades are not so numerous. The first blade arrangement has a relative pitch, i.e. the ratio of the blade pitch at the wheel entrance to the radial length of the blade  $t_1 =$

$\frac{t_1}{r_1 - r_2}$ , about  $t_1 = 0.12$ , the second has a relative pitch about  $t_1 = 0.6$ .

As to the efficiency both arrangements, as experience has shown, are of equal value, however, from the standpoint of construction, arrangements with less numerous blades are more convenient. In addition theoretical considerations imply that the number of blades of the inward flow expander wheel may be less than that recommended for a centrifugal compressor wheel. That is why at present we produce wheels with a blade pitch of about  $t_1 = 0.6$ .

The investigations performed have shown that in most cases the least energy losses are peculiar to those constructions where the inlet angle of the wheel blade is  $90^\circ$ .

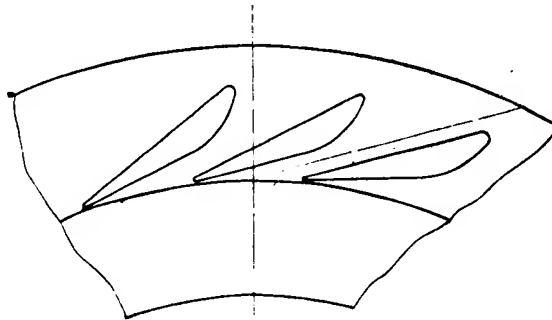


Fig. 5. Nozzle arrangement.

The stationary nozzles are shaped approximately like those shown in Fig. 5. The nozzle angle is usually  $14-20^\circ$ . The velocity factor for nozzles of this type, that is the ratio of the actual discharge velocity  $C_1$  to the isentropic discharge velocity is  $\phi = 0.96 - 0.975$  if the discharge velocity is close to the sonic, i.e. at  $C_1 = 1$ . If the  $\frac{C_1}{a_{cr}}$  ratio decreases  $\phi$  is lowered, that means that the losses through the nozzles increase. The negative influence of the section in the nozzles may chiefly account for this.

Calculating the stationary nozzles for a low-temperature expansion turbine one should take into account the deviations of real gas from the ideal gas laws by introducing the factor for the supercompressibility of gases  $Z = \frac{PV}{RT}$ . This factor is considered to be constant in the course of expansion of the gas in the nozzles.

Reaction expansion turbines have no devices to control the capacity. Therefore the oxygen plants have two identical turbines. During the starting period both of them are in operation. At liquefying conditions one of them operates and the other remains in reserve. Sometimes it is necessary to change the nozzle box, for instance, one set is designed for winter conditions and the other for summer. However, the design of a controllable reaction expansion turbine is also possible. Preliminary experimental data have shown that it is possible to control the capacity over a relatively wide range, the efficiency being lowered comparatively little.

Besides the design of controllable turbines, our task is to develop radial expansion turbines with two or even more expansion stages.

Behendt bogtryk, Denmark

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## The Main Trend in the Designing of Large Gaseous Oxygen Plants

Les tendances principales dans la construction de grandes installations pour oxygène gazeux

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*SOMMAIRE. Les grandes installations modernes pour la production d'oxygène gazeux sont toutes basées sur le principe de basse pression en utilisant des turbomachines. Le travail sur le développement d'appareils pour la rectification de l'air fut commencé dans l'Union soviétique après 1940 sous la direction de l'Académicien P. L. Kapitsa en se basant sur sa turbine à expansion hautement effective.*

*A la suite de vastes recherches et du travail expérimental exécuté à VNIIMash un certain nombre d'installations furent construites dont les capacités variaient entre 3600 et 15000 m<sup>3</sup>/hr d'oxygène gazeux.*

*Dans le rapport le circuit de telles installations est examiné et des solutions concrètes sont présentées en ce qui concerne les problèmes techniques qui se posent à la suite de la nouveauté des projets et des grandes dimensions des installations.*

*La construction des colonnes de rectification et des évaporateurs-condenseurs à long tube est montrée. La solution des principaux problèmes de construction se rapportant au drainage de l'air de vapeur d'eau et d'anhydride carbonique est discutée. Le gel du régénérateur est empêché par un régénérateur d'azote supplémentaire avec bouillage ordinaire. La méthode pour dégeler des unités à basse pression est décrite.*

*L'expérience acquise pendant l'opération des installations à grande échelle a démontré que la construction proposée assure leur opération sans devoir dégeler pendant des périodes allant jusqu'à deux ans.*

The main current trend in the development of technological processes for the production of gaseous oxygen rests on the low pressure principle utilizing turbomachines.

All relatively large plants were built until late on a two-pressure cycle. The introduction of high pressure made it possible to remove from the regenerators by a reverse flow the moisture and carbon dioxide that remained on the packing after the direct air flow. The high pressure provided also the production of the greater part of refrigeration. The so-called rectification process reserves available in these plants were also used for the production of refrigeration by expanding in the expansion turbine a part of the nitrogen taken from under the condenser cover.

Plants, operating in two-pressure cycle, are sufficiently reliable and maneuverable but complex in their equipment and in operation due to the presence of high pressure air, piston compressors, equipment for the chemical purification of air of carbon dioxide and an ammonia refrigerating system.

To eliminate the high pressure and to operate with only low pressures it was necessary to design a highly efficient expansion turbine that would allow to compensate refrigeration losses with a minimum worsening of the rectification process. It was also necessary to develop a system of heat exchanging devices for freezing out the moisture and carbon dioxide present in the air and their subsequent complete removal by a reverse flow.

P. L. Kapitsa suggested in 1939-1940 a new highly efficient type of expansion turbine - a reactive radial type turbine. The new machine proved to be reliable in operation and to have an efficiency of over 80%.

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Work was started simultaneously under the guidance of P. L. Kapitsa on the development of the first air separating plants operating only with low pressures. This work actually laid the foundation for modern methods of producing large amounts of gaseous oxygen. However, no method was found at that time able to provide continuous operation of the plants for a long period.

Heat-exchange devices, designed in the USA in the form of regenerators - recuperators with an unbalanced flow, are very complex to manufacture and not efficient enough (due to a large resistance and a high temperature difference on the warm end of the regenerator).

Further investigations resulted in the development of a more simple in design and reliable method of purifying air of water vapour and carbon dioxide with clogging of the regenerators eliminated. The problem was solved by including an additional nitrogen regenerator with ordinary packing in the layout.

The essence of this solution can be demonstrated by comparing it with the method providing non-clogging of the regenerators in two-pressure plants.

Conditions providing prolonged operation of the regenerators of two-pressure installations without being clogged with carbon dioxide are created, as is known, due to a surplus (by 3 to 4%) amount of the reverse flow. This part of the reverse flow is provided by high pressure air purified of carbon dioxide and water vapour in special apparatus. Thus, in regenerators, packed with discs of corrugated aluminium band heat-exchange conditions are obtained providing, during the period of cold blowing, a complete sublimation of all the admixtures remaining on the packing after the air has passed and their removal from the regenerators.

A surplus reverse flow is also used in oxygen regenerators of low-pressure plants. In nitrogen regenerators the relation of the reverse flow to the direct one becomes in this case even less than unity. The conditions of heat-exchange in nitrogen regenerators under which the air is cooled to a state of dry saturated vapour with the accumulation of carbon dioxide in the ports of the packing eliminated are created by providing an unbalanced so-called "loop" flow. This flow allows transfer of part of the heat load of the nitrogen regenerators to the nitrogen subcooler and heater thus reducing the temperature difference at the cold end of the regenerators and facilitating the sublimation of the carbon dioxide remaining on the packing after the warm flow and its removal by the nitrogen.

A specific feature of this solution, as it has been mentioned, is the unbalanced flow provided by the installation of a third nitrogen regenerator.

The sequence of the flows through each nitrogen regenerator is as follows:

1. air is passed in a direct flow and cooled while the moisture and carbon dioxide, present in the air, precipitate on the cold packing;
2. nitrogen is passed cooling the packing and removing the deposited impurities;
3. a flow of air (a "loop" flow) is passed in the same direction as nitrogen and additionally cools the packing in the lower (cold) part of the regenerator and

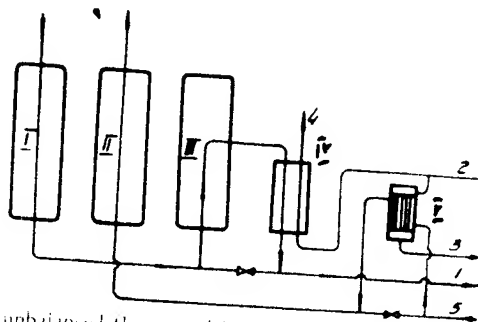


Fig. 1. Diagram of unbalanced flow provision system. I, II, III-nitrogen regenerators, IV-heat exchanger for air passing to turbine; V-nitrogen heater. Direction of flows - 1. Air to lower column; 2. Air from lower column; 3. Liquid air to lower column; 4. Air to expansion turbine; 5. Nitrogen from subcooler.



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is then discharged at 160-180°K through special valves from the middle part of the regenerator.

The flows are passed through the other nitrogen regenerators in the same succession and while a direct flow of air passes through the first (Fig. 1) regenerator, nitrogen flows through the second and the "loop" air flow is run through the third one.

The mean temperature difference at the cold end of the regenerators is maintained within 5 to 6°C by changing the heat load of the nitrogen heater as well as the amount of "loop" air.

Thus the purification of the entire amount of air, supplied for separation, of moisture and carbon dioxide is performed in the regenerators.

An additional purification of the unbalanced flow, which is required in the case of a disturbance of temperature conditions in the regenerators, is performed as this flow is cooled in a heat-exchanger by means of a heat exchange with the air passing from the lower column to the turbine.

The detaining of the carbon dioxide crystals carried along by the air from the regenerators or heat-exchanger as well as the reduction of acetylene content in the air flow passing through the expansion turbine is provided by washing the air on three washing plates arranged in the lower part of the lower column.

By applying the above new solutions on the expansion turbine design and the prevention of clogging of the regenerators, a technological process was elaborated which constituted the basis of a number of low pressure air separating units with capacities of from 5,600 to 12,500 - 15,000 norm. m<sup>3</sup>/hr of gaseous oxygen. The largest air separating unit with a nominal capacity of 12,500 norm. m<sup>3</sup>/hr of tonnage oxygen is manufactured on a serial scale. The first experimental unit has been in operation since 1986 and worked for the first two years without stopping.

According to the data of tests the specific power consumption was from 0.43 to 0.45 kW/hr per norm. m<sup>3</sup> of oxygen at a nominal capacity of 12,500 norm. m<sup>3</sup>/hr at 20°C and 760 mm Hg) when operating without the krypton and pure oxygen block and at a value of isothermal efficiency of the turbo-compressor amounting to 60%.

The schematic technological diagram of the part of this unit connected with the production of tonnage oxygen is given in Fig. 2.

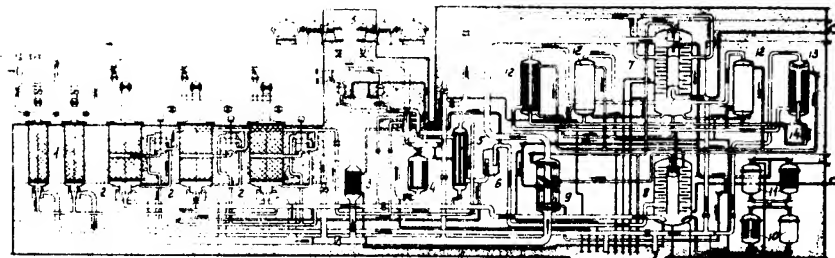


Fig. 2. Flow diagram of a 12,500 m<sup>3</sup>/hr oxygen plant. 1. Oxygen regenerator; 2. Nitrogen regenerator; 3. Heat exchanger - nitrogen heater; 4. Acetylene adsorber; 5. Heat exchanger for air passing to turbine; 6. Liquid separator; 7. Upper rectification column; 8. Lower rectification column; 9. Liquid nitrogen and air subcooler; 10. Carbon dioxide filter; 11. Acetylene boiler; 12. Condenser; 13. Condenser evaporator of produce oxygen; 14. Acetylene adsorber; 15. Expansion turbine unit.

The main air separation block may be equipped, if necessary, with additional equipment for the production of primary krypton concentrate and the required amount of pure oxygen (99.5% O<sub>2</sub>). Fig. 3 illustrates the flow diagram for obtaining these products of air separation.

Additional equipment is installed in some cases for the production of a certain amount of pure nitrogen.

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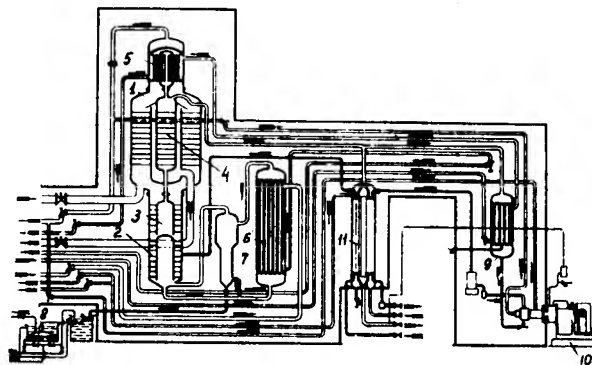


Fig. 3. Flow diagram of additional equipment for the production of krypton and pure oxygen. 1. Concentration part of krypton column; 2. Distilling part of krypton column; 3. Pure oxygen column; 4. Column for washing pure oxygen of krypton; 5. Krypton column upper condenser; 6. Condenser-evaporator; 7. Liquid separator; 8. Krypton concentrate evaporator; 9. Pure oxygen condenser-subcooler; 10. Liquid oxygen pump; 11. Pure oxygen heat exchanger.

A respective amount of air is taken from the upper part of the regenerator during the "loop" flow of air for heat exchange with the primary krypton concentrate and pure oxygen or nitrogen.

In designing a plant of a capacity mentioned above, new engineering solutions were required for the condensers, rectification column, insulation and other elements of the unit as the existing solutions were unfit or caused serious complications in operation.

As a result of investigating various models, a condenser was selected with the oxygen boiling inside the pipes and nitrogen condensing in the interpipe space. The application of this type of condenser made it possible to use considerably longer (up to 5 metres) pipes without any essential increase of the hydrostatic temperature depression and to provide the necessary surface with a considerably smaller number of pipes at a relatively small diameter of the shell. A change of the character of the load on the tube sheet caused by the transfer of the oxygen boiling process inside the pipes, made it possible to reduce the thickness of the tube sheet by three times and to simplify the construction of the condenser in general.

One of the main factors limiting the possibility of having the upper and lower columns each as a single apparatus was too large a load upon the drain device. After experimental investigations of the hydraulics of the plates, a double-drain meshed circular plate was developed which considerably reduced the load upon the drain device. This type of plates allowed to increase by 1.5 times the velocity of the vapour flow in the columns almost without any increase of resistance and, consequently, to considerably reduce the dimensions and weight of the apparatus per unit of processed air as well as to design the columns in the form of one apparatus each.

A decrease of the load upon the drain device allowed to provide also reliable and efficient operation of the rectification columns within a wide range of capacity variations (the ratio of the maximum capacity to the minimum amounting to 2).

Unlike the ordinary method of insulating apparatus located inside a shell by filling the entire inner space with insulating material, large units were made jacketed with insulating material only between the walls of the jacket. This method of insulation reduces the insulating material requirements and the apparatus and lines become accessible for assembly, repairs and inspection without the removal of the slag wool. Besides, the duration of the starting period is reduced. This is why the above method of insulation was considered most rational for large units notwithstanding the somewhat greater losses into the surrounding medium.

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With the flow of high pressure air purified of moisture and carbon dioxide excluded from the layout a new defrosting system of the plant was required. The plant is defrosted by air that had been passed through the regenerators and heated up to 20 - 30° C. in a special heater. The heated air is distributed among the apparatus of the separation block to be defrosted and then passed in a reverse flow through the regenerators. With such an arrangement of the flows the regenerators are the coldest apparatus throughout the entire defrostation time. Thus, during this process all impurities accumulated in the apparatus in the course of their operation are removed.

An increase in the size of the plants required a new solution of the problem of control. For most efficient operation the plants are equipped with recording control and measuring instruments and remote control devices. The latter are mounted on all the main pipelines thereby allowing more convenient arrangement of the apparatus and simplification of the lines. The remote control is performed from the central panel.

A system of automatic devices is being introduced at present that will make it possible to automatically maintain preset operating conditions without any interference of an operator.

A complex automatic system is being developed for the solution of a larger problem of automatically setting and maintaining the operating conditions in accordance with the load.

The main engineering solutions described above and tested on plants already manufactured are used as the basis for developing still larger oxygen plants and units for complex separation of air.